50X1-HUM

Page Denied

Aerodynamic Supports For High Speed Motors and Turbines

Dr Techn. Science S. A. Sheinberg
Cand. of Tech. Science A. M. Kharitonov

The fundamental obstacle to the wide spread industrial application of high speed machinery are the shortcomings of bearings now in use.

In rolling contact supports an increase of speed causes progressively increasing dynamic forces which lead to extremely stringent requirements on the materials of the bearings and on the requirements on the accuracy of their manufacture. Even with strict application of these factors, endurance life of rolling bearings fall sharply with increase of speed, which is measured by the product Dn, where D - diameter of the arbor seat in MM and n - number of revolutions per minute. Practically speaking, the useful values of Dn do not exceed 500 thousand for rolling contact bearings of higher quality classifications, which corresponds to a circumferential speed on the arbor seat of 25 meters per second.

In arbor supports having hydrodynamic liquid lubricated bearings, the situation is more favorable inasmuch as spheres subjected to heavy centrifugal loads are eliminated. These are replaced by a lubricant film, whose limiting factor is the viscosity of the lubricating liquids.

As is known, the frictional moment M in the lubricant layer of the sliding bearing is approximately

$$M = \frac{\eta S v r}{\delta}$$

where η - absolute viscosity of the lubricant

S - area of the working surface of the bearing

v - circumferential speed of the rotating shaft

δ - average clearance, that is the difference of radii of the bearing and the shaft ($\delta = r_B - r_{SH}$).

From the formula it is apparent that losses due to friction can be reduced by reducing the (sic) dimensions of the bearing since this reduces the speed, or by lowering the viscosity of the lubricant. Concerning the average clearance, an increase in its magnitude is impossible due to requirements of stable rotation and prevention of vibration.

The practicability of the first method of reducing frictional losses is limited by the demands of rigidity of the shaft, which falls rapidly with reduction of shaft diameter. Therefore, for liquid lubrication, even using the least viscous lubricants (kerosene or water), the product Dn is limited to a value of the order of 200,000, which corresponds to a circumferential velocity of 10 meters per

second. Further increase of D_n is successful only by introducing special precautions, which completely complicate the design of the machine and its use or by reducing the working surface by means of cutting out pockets resulting in turbulence of the lubricant with an accompanying requirement for intensive water cooling. This causes a requirement for complicated sealing.

The second method - increasing the speed of operation by means of using gases for lubricants of low viscosity - is far more practicable.

The application of aerodynamic supports - both radial or thrust bearings having gas and particularly air lubrication, radically solves the problem of providing the long life desirable in the working supports for high speed machines.

Aerodynamic supports enable practically any circumferential velocity to be attained on the shaft surface (D_n = 2,000,000 and more), completely fulfilling the requirement of design rigidity of the shaft because of its maximum possible diameter. The service life of these supports is practically unlimited, presuming that they are properly designed and manufactured. As a result of the fact that viscosity of air is many times less than the viscosity of oil, water, kerosene, and other liquids used for lubrication; the loss of power and heat generation in aerodynamic bearings are so small, that there are no requirements whatever for increasing heat flow from the support - normal ventilation is sufficient.

It is essential to note that the temperature dependence of liquids and gases are directly opposite - the circumstance of increasing temperature causes a sharp reduction of viscosity of liquids, gases somewhat increase in viscosity - true, to a lesser degree.

Thus the viscosity of air at a pressure of 1 Kg/cm^2 and temperature of -194° and 229°C are correspondingly 0.00000056 and 000 00269 Kgm^{-2} sec.

The comparatively insignificant effect of temperature on the viscosity of gases enable broad possibilities for the application of gas lubrication under conditions of low and high temperatures. There are, for instance, reports that bearings with argon lubrication have successfully operated at 2200°C.

Increase of pressure similarly insignificantly affects the viscosity of the gas lubricant. Thus at a temperature of 20°C and a change of pressure from 1 to 100 Kg/cm^2, the viscosity of air correspondingly is 0.00000180 and 0.00000199 Kgm^{-2} sec.

As the pressure in the lubricant film of aerodynamic supports, in general, does not exceed 10 Kg/CM^2 then it is practical to consider that viscosity is independent of pressure.

Work on high speed electromotors on aerodynamic supports was begun in ENIMS back in 1950.

The purpose of this work was embodied in the development of electric spindles for grinding small diameter openings, at speeds of rotation of the order of 30 - 50,000 RPM.

Testing of the initial examples of electrospindles underscored the basic practicability of solving the above discussed problem. Simultaneously it was demonstrated that reworked equipment was unsatisfactory. The primary difficulties were the following:

a) The increase of rotor vibration of the rotor with increase of rotor speed which led to scoring of the working surfaces of the aerodynamic supports which were made of quenched steel in the first examples.

b) Insufficient accuracy of coaxiality of the bearings.

c) A high friction moment on the shaft of the electric spindle from the surface of the stationary bearing at start up.

d) Poor accuracy of dynamic balance of the rotor.

As a result of extensive work by the authors, in 1957 there was established a construction of a high speed electric motor for application in electric spindle drives (announced in authors communication No. 579255/25) in which the earlier limitations were overcome and the solution of a series of more important problems associated with aerodynamic supports were given.

The electro spindle of the type ЭШ -19 (fig. 1) was designated for grinding bores to 25 mm. It is a 3 phase asynchronous short circuited electric motor with following technical specifications:

power - 1 kvt, 220 volts, amperage -3 a;
0.85, k.p.d. -80%, speed of rotation -
48000 synchronous rpm; diameter of shaft 32 mm.

At synchronous speed, the circumferential velocity at the rotor surface is 124 meters per second and on the bearing seats - 80 meters per second. The Dn product is equal to one and one half million, which is three times the analogous parameter for ball bearings.

The shaft 1, with the pressed on rotor 2, rotates in two cylindrical bearings 3, located axially in supporting conical surfaces of the front and rear end bell of the motor 4. The bearings are pulled up by means of threaded collars 5 and 6. The end bells are assembled into the shell 7 on accurate locating surfaces and are fastened in place by screws 8. On the end surface of the extension of each end bell is installed the thrust washer 9. The accurate correction of coaxiality of the two bearings is achieved by means of the screws 10.

Into the lubricant clearances of both bearings air or some other gas under pressure is introduced through the inlet 11 and distributive tubing 12.

The single direction of gas feed into the bearing applies a load on the shaft which damps its vibration on the air supporting pad which otherwise occurs at high speeds. These vibrations, called half speed whirl, are the result of rotation of the shaft in the bearings with small relative eccentricities, under which the location of the shaft axis becomes instable. The fixed direction of radial load forces the axis of the shaft into the zone of maximum eccentricity, corresponding to its stable position. This eliminates half speed whirl. The compressed gas impinges on the loaded upper portion of the bearing where the pressure is somewhat less than atmospheric. Increase of pressure in that region significantly increases the carrying capacity of the bearing. At a pressure of 3 atmos. and minimum clearance $h = 4$ microns, the carrying capacity of the bearing (diameter 32 millimeters, length 55 millimeters) is equal to 85 kilograms, when under a pressure of one atmosphere the same bearing can carry a load of 30 kilograms.

The compressed gas escaping from the bearings in the axial direction establish a load bearing air film which supports an axial load. This occurs in the following manner: The gas passing through the lubricating clearance, collects under the threaded ring 6 and loads against the end surface of the shaft, pressing the rotor against the thrust bearing at the left. Simultaneously these surfaces continue to be divided by the gas lubricant pad, formed as the result of efflux of gas, entering the left bearing. Such construction enables accurate axial location of the rotor, i.e., a means for reducing the requirement of accurate correspondence of the distance between thrust bearings and rotor length.

Increasing the pressure in the lubricant film of the bearing prevents the possibility of entry of abrasive dust or particles, etc. This reduces the requirement of protecting the bearings by some kind of sealing, simplifies the construction and enables use of the described motor in atmosphere conditions which are contaminated by corrosive gases, dust and abrasives which, in particular, is very important for grinding heads.

The most important condition for normal operation of the rotor on aerodynamic bearings is the coaxiality of both rotor bearings. A non coaxiality over the length of each bearing should not exceed 2-3 microns. In view of this, to provide such a high accuracy under rigid assembly of bearings has presented, up till now, tremendous difficulties in known constructions of aerodynamic supports used in self acting bearings assembled into pivots or in flexible membranes. Similar constructions work well only at comparatively low speeds (to 20,000 RPM). With an increase of speed such a non rigid support of the bearings enablesoccurrence of vibration. They are also inappropriate for transmitting significant radial loads and cannot transmit axial loads. The thrust bearing in that case must be installed into an independent Cardan support.

In the above described motor, the bearings are rigidly installed in end bells, but the extensions in which the bearings are installed is associated with the thin rim of the end bell by a thin ring which can deform as a result of tightening the screws 10. This enables the possibility of rotating the axis of each bearing relative to the axis of the motor housing and, therefore, accurate correction of the coaxiality of both bearings, completely compensating for errors in manufacturing the motor parts.

The rigidity of the installation of bearings enables application of the simplest thrust bearings, using the gas consumed by radial bearing operation. Simultaneously achieving accurate location of the rotor in an axial direction by means of single direction application of gas pressure on the open shaft end.

The verification of coaxiality of the bearings is accomplished by an accurate cylindrical insert which closely fits the bearings. As a result of the cumulative errors of manufacture of parts of the motor (motor frame, end bells, inserts) the clearance between the end surfaces of the installed surfaces vary around the circumferential direction, as is shown schematically on fig. 2. Thanks to its short length, the finished cylindrical band does not hinder such axial divergence. Let ΔA - maximum clearance around the circle Moving the scale and accompanying fastened end bells relative to the motor frame, we can establish this clearance equal for instance 0.1 mm, and then with the aid of the screw Б make equal (by means of a compensating element) clearances Δ Б and ΔA. With the aid of three or more regulating screws located around the circumference of the end bell it is easy to obtain an adjustment of constant clearance around the circle to an accuracy of 0.005 mm. For a relation $d/l = 2.5$ this corresponds to a coaxiality error over the length of the bearing of no more than 0.002 mm.

By the same method the second bearing is adjusted.

In the final adjustment, the insert is removed, the motor is assembled and the toroidal locating surfaces of the end bells and the body (frame) are tied together by bolts as shown on figure 1. The shortcomings of known constructions of aerodynamic bearings are the impossibility of regulating the bore of the bearing, which hinders attainment of required clearance and requires bearing change even after insignificant damage to its working surface.

In the electrospindle ЭШ-19, the exterior surface of the bearing is made in conical form. Thanks to the fact that the bearing is made of material having significant elasticity, its exterior conical surface deforms under tightening axially, compensating for errors of manufacture of contacting conical surfaces. The final turning of the cylindrical working surface occurs after assembly with the end bell. Later the diameter of the bearing can be reduced by proper tightening of the threaded ring, during which the geometric accuracy is not lost, since the motion along the axis is not great and the contacting cones are always in contact.

At present the Moscow Electrode Works manufactures for aerodynamic supports graphitic materials Д and E , which are improved versions of materials АУГ-3 and АУГ-4 and having improved mechanical and antifriction properties according to Inst. of Machine Development AN USSR. (2).

Impregnating these materials by metals (lead, babbet, cadmium and others) prepared by Moscow Electrode Works, almost two-fold increased their strength, somewhat reduced brittleness and eliminated porosity, which was one of the primary impediments to establishing the air film between the shaft and insert.

In the electrospindle ЭШ-19, supports are of graphitic materials, designation E, having a higher heat conduction than Д material, better antifriction properties and lower wear.

Coefficient of dry friction of this material depends on the quality of shaft surface finish and insert and for speeds to 10 meters per second is in the range 0.04-0.05.

For aerodynamic supports in addition to a low coefficient of friction, the non-scoring or welding of graphite with metals during contact at high sliding speeds is extremely important. During application of inserts of steel, bronze, iron-graphite and other materials, dry friction occurring as a result of inaccurate manufacture and selection (for example distortion of inserts) resulted in serious damage of working surfaces of shaft and inserts even to welding. Application of graphite removed such occurrences. In the worst case the problem is limited to damage to the insert, which is easy to change.The shaft remains satisfactory for further use. Steel P-9 quenched to 58-60 R_c has been used for shafts. To reduce dimensional change during use the quenched parts are cold treated in liquid nitrogen.

Dynamic balance is extremely important in achieving stable non-vibrating operation at high speed of rotors. This is equally true of rotors operating in aerodynamic bearings.

The minimum air space h_o between shaft and supports at limiting load can be reduced to 3 microns. Considering that the relative motion of the axis of the rotor under the effect of imbalance should not exceed 0.15 h_o , we obtain a maximum allowable motion of approximately 0.5 microns.

As is known

$$M_\partial = Ge = rg$$

where M_∂ - moment of remaining imbalance

G - weight of rotor

e - divergence of physical and geometric axes of rotor under effect of imbalance

r - radius at which material is removed

g - weight of removed metal

In our instance (for $G \approx 1$ kg, $e = 0.5$ microns, $r = 1.5$ cm) the balancing machine must respond to a load $g = 0.32$ gram.

For achieving these requirements a special rig was designed and constructed (fig. 3), in which a mechanical indicator is used. The rig is quite simple and can be used in any machine construction plant. The mechanical indicator, exhibiting low sensitivity, was replaced with a piezoelectric pickup from a tonearm. The balance rotor 1 is mounted on sliding bearings similar to those in the electrospindle (in view of the low speed of rotation they are lubricated by kerosene) disposed in the frame 2 such that one of its end surfaces is located in the plane of fastening the frame on knife blades, the second moves with the framework.

The vibration of the framework is transmitted as an angle to the quartz pickup 4 by means of the crank arm and by means of the amplifier 5 is applied to the oscillograph 6.The pickup is pressed against the crank arm 3 by a weak spring.

The magnitude and location of imbalance is determined by means of selection and relocation of balancing weights.

For reducing extraneous effects, the machine is supported on four springs 7 in assembly 8.

The speed of rotor rotation during balancing is 3 - 4000 RPM.

In the head end of the balancing machine there is a special electric motor 9 . With the purpose of maximum reduction of rotor unbalance the eletromotor rotor is solid. The support bearings of the electric motor are of the sliding type of antifriction graphite with adjustable bore diameter. The sheave of the electric motor is finish machined after installation on the shaft as a result the eccentricity of the sheave is held to 0.01 mm.

These measures lead to a minimum of vibration of the driving motor and its effect on the rig.

The transmission from the drive to the balance rotor is achieved by a belt of chlorovinyl tube which is heat welded.

The accuracy of balance achieved on the rig is less than 0.02 grams centimeters which meets requirements discussed above.

In conclusion it can be said that the high speed motor on aerodynamic support bearings has been successfully tested under laboratory conditions as an internal grinding head. This head enabled high quality grinding - during finish grinding to 13th class of surface finish - and higher productivity. This is explained initially on the basis of the greater bearing shaft diameter and therefore high shaft stiffness. The grinding head is simple and desirable for production.

In the process of manufacturing aerodynamic supports of the motor the question is of construction and support of the stationary bearing parts in the end bells, accurate verification of coaxiality of stationary bearing parts, entering into their load carrying capacity, damping shaft vibration on the air bearing support, axial location of the rotor, protection of bearing against dirt contamination and selection of materials for shaft and stationary bearing part.

Founded on this basis, aerodynamic bearings and technology of their manufacture can be successfully applied in all kinds of high speed motors and machinery.

LITERATURE

1. Sheinberg, S. A. Gas Lubrication of Sliding Bearing (Theory and Calculation) Trenie Iznos

Mashinakh Sb. VIII, IZD A.N. USSR, 1953

2. Plutalova, L. A. Graphitic Sealing of Rotating Shafts Vestnik Mashinostroeniya No. 2, 1957

	C	W	Cr	V	Mo
P-9	0.85-0.95	8.5-10	3.8-4.4	2-2.6	0.3

Page Denied

Next 3 Page(s) In Document Denied